Amendments to the Specification:

Please amend the specification as follows:

Please replace the paragraph starting at page 1, line 7 with the following:

In recent years, to meet demands for increased shift comfort, improved driveability, and reduced fuel consumption and exhaust emissions, there have been proposed and developed toroidal continuously variable transmissions often abbreviated to "toroidal CVTs", in which a transmission ratio is steplessly variable within limits. One such toroidal CVT has been disclosed in Japanese Patent Provisional Publication No. 10-331938 (hereinafter is referred to as "JP 10-331938"), corresponding to United States Pat. No. 6,030,309. On such a toroidal CVT as disclosed in JP 10-331938, engine power (output torque) is transmitted from an input disk to an output disk via a traction oil film formed between a power roller and each of the input and output disks, using a shearing force in the traction oil film at high contact pressure. The input and output disks coaxially oppose each other. The toroidal CVT has a trunnion serving as a power roller support that rotatably supports the power roller, which is interposed between the input and output disks and is in contact with a torus surface of each of the input and output disks under preload. During transmission-ratio changing of the toroidal CVT, in order to obtain a desired transmission ratio determined based on the magnitude of a gyration angle of the power roller, the power roller is shifted from a neutral position, at which a rotation axis of the power roller intersects the center of rotation (common rotation axis) of the input and output disks, by slightly shifting or displacing the trunnion in a direction of a trunnion axis perpendicular to the rotation axis of the power roller via a servo piston of a hydraulic servo mechanism that operates in response to a hydraulic pressure generated by a prime-mover driven oil pump that is constantly driven by a prime mover (an engine) during operation of the ~~primer~~ <u>prime</u> mover. In more detail, during a forward running mode the hydraulic pressure is directed to the hydraulic servo mechanism via a forward ratio control valve, whereas during a reverse running mode the hydraulic pressure is directed to the hydraulic servo mechanism via a reverse ratio control valve. By virtue of a side slip force occurring in a very limited contact zone between the power roller and the input and output disks due to the slight offset or slight vertical displacement of the power roller, the power roller is self-tilted or self-inclined. Owing to the self-inclining motion of the power roller, a

first diameter of a circular-arc shaped locus drawn by movement of the very limited contact point between the power roller and the output disk on the torus surface of the output disk and a second diameter of a circular-arc shaped locus drawn by movement of the very limited contact point between the power roller and the input disk on the torus surface of the input disk, that is, a ratio of the first to second diameter can be continuously varied, thus continuously varying a transmission ratio. Generally, in the toroidal CVT, a degree of progress for transmission-ratio changing is fed back to the hydraulic servo mechanism, so that the trunnion gradually returns to its initial position as the transmission-ratio changing progresses. When the gyration angle based on a desired transmission ratio corresponding to a transmission-ratio command signal value has been reached, the vertical displacement of the trunnion is returned to zero, so as to terminate the inclining motion of the power roller, and to attain the return of the power roller to neutral, and thus to maintain the desired transmission ratio corresponding to the ratio command signal value.

Please replace the paragraph starting at page 3, line 19 with the following:

When the output disk is driven by road wheels due to back-flow of torque from the road wheels to the output disk, as a push-back force or a reaction force from a contact portion between the power roller and the input shaft, the power roller, interposed between the input and output disks under preload, receives a component force acting in the trunnion-axis direction. This causes a slight offset of the power roller from its neutral position in the trunnion-axis direction that upshifts ~~up-shifts~~ the toroidal CVT to a higher transmission ratio. As a result of this, owing to the self-inclining motion of the power roller, the upshift of the toroidal CVT to a higher transmission ratio undesirably occurs ~~undesirably~~. If the ~~primer~~ prime mover is restarted and the vehicle is accelerated from standstill on the assumption that the toroidal CVT has been undesirably upshifted to a high transmission ratio owing to hauling or coasting in the stopped state of the prime mover, there are the following drawbacks.

Please replace the paragraph starting at page 5, line 22 with the following:

In order to accomplish the aforementioned and other objects of the present invention, a toroidal continuously variable transmission comprises an input disk to which rotation of a prime mover is transmitted, an output disk coaxially arranged with and opposed to the input

disk, the output disk adapted to have a driving connection with and to have a driven connection with a road wheel, a power roller interposed between the input and output disks under axial preload for power transmission, a trunnion rotatably supporting the power roller to permit a tilting motion of the power roller about a trunnion axis perpendicular to a rotation axis of the power roller for ratio changing, a hydraulic servo mechanism connected to the trunnion to move the trunnion in a direction of the trunnion axis so as to cause the tilting motion of the power roller by creating an offset of the trunnion from a neutral position in the direction of the trunnion axis, the neutral position being a non-ratio-changing position at which the rotation axis of the power roller intersects a common rotation axis of the input and output disks, a primary oil pump driven by the prime mover to produce a hydraulic pressure, a secondary oil pump driven in response to rotation of the road wheel to produce a hydraulic pressure, the hydraulic servo mechanism hydraulically operated by either of the hydraulic pressure from the primary oil pump and the hydraulic pressure from the secondary oil pump, a ratio-change-control hydraulic system that supplies the hydraulic pressure discharged from the secondary oil pump to the hydraulic servo mechanism to prevent the offset of the trunnion in the trunnion-axis direction, corresponding to an upshift, occurring owing to rotation of the road wheel in a stopped state of the prime mover, and the ratio-change-control hydraulic system by which a modulated hydraulic pressure is constantly produced and outputted to the secondary oil pump during operation of the ~~primer~~ prime mover to hold the secondary oil pump at an inoperative state during the operation of the prime mover.

Please replace the paragraph starting at page 8, line 8 with the following:

Referring now to the drawings, particularly to Fig. 1, a toroidal continuously variable transmission (abbreviated to "toroidal CVT") of the embodiment is exemplified in a half-toroidal continuously variable transmission combined with a lock-up torque converter 2. In the power train for the toroidal CVT of the embodiment shown in Fig. 1, engine torque (driving torque) is transmitted from an engine 1, serving as a prime mover, via lock-up torque converter 2 to a forward and reverse changeover mechanism (F/R changeover mechanism) 3. F/R changeover mechanism 3 functions to transmit input rotation to an input shaft or an input disk of the toroidal CVT without changing a direction of rotation in a drive range (D range) of a forward running mode. F/R changeover mechanism 3 also functions to transmit input

rotation to the toroidal CVT input shaft while changing a direction of the input rotation in a reverse range (R range). That is, F/R changeover mechanism 3 reversibly transmits the input rotation of the prime mover to the input disk. F/R changeover mechanism 3 further functions to shut off power transmission to the toroidal CVT input shaft in a parking range (P range) or a neutral range (N range). F/R changeover mechanism 3 is generally comprised of a planetary gearset, a forward clutch, and a reverse brake. At the subsequent stage of the F/R changeover mechanism, a front toroidal CVT mechanism (or a first variator unit) 4 and a rear toroidal CVT mechanism (or a second variator unit) 5 are set in tandem and coaxially arranged in the interior space of the toroidal CVT casing in a manner so as to construct a so-called "double cavity type toroidal CVT". First and second toroidal CVT mechanisms 4 and 5 have the same ~~in~~ construction. First toroidal CVT 4 is comprised of a pair of input and output disks 6 and 7 coaxially arranged and opposing each other, a pair of power rollers (8, 8), and a power roller support or a trunnion 12 (describer later). Each of input and output disks 6 and 7 has a torus surface. Power rollers (8, 8) are interposed between input and output disks 6 and 7, such that power rollers (8, 8) are in contact with the torus surfaces of input and output disks 6 and 7 under axial preload. Power rollers (8, 8) are symmetrically arranged to each other with respect to a main torque transmission shaft 9. First and second CVT mechanisms 4 and 5 are arranged in reverse to each other on main torque transmission shaft 9, such that the output disk included in first toroidal CVT mechanism 4 and the output disk included in second toroidal CVT mechanism 5 are opposed to each other with respect to an output gear 11 fixedly connected onto a cylindrical hollow output shaft 10. Although it is not clearly shown in Fig. 1, of two input disks (6, 6), the input disk included in first toroidal CVT mechanism 4 is preloaded axially rightwards (viewing ~~Viewing~~ Fig. 1) by means of a loading cam device (not shown). The loading cam device is designed to produce a magnitude of the axial preload substantially proportional to input torque transmitted from lock-up torque converter 2 via F/R changeover mechanism 3 to the toroidal CVT input shaft. On the other hand, the input disk included in the second toroidal CVT mechanism 5 is permanently biased axially leftwards (viewing Fig. 1) by way of a spring bias. Each of input disks (6, 6) is supported on main torque transmission shaft 9 by way of ball-spline-engagement, so as to permit each of input disks (6, 6) to axially move relative to the main torque transmission shaft, and to rotate about

the main torque transmission shaft. Output disks (7, 7) and cylindrical hollow output shaft 10 are integrally connected to each other or integrally formed with each other, and coaxially arranged with each other. Output ~~Out-put~~ disks (7, 7) are linked to output gear 11 via cylindrical hollow output shaft 10 by way of spline engagement. In contrast to input disks (6, 6), each of output disks (7, 7) is axially stationary. Thus, output gear 11 is rotatable relative to main torque transmission shaft 9.

Please replace the paragraph starting at page 14, line 26 with the following:

As can be seen from the cross sections illustrated in Figs. 2A, 2B, and 2C, in the toroidal CVT of the embodiment, secondary oil pump 22 is constructed as a reciprocating plunger pump that is comprised of an eccentric cam (or a pump driving element) 24 and a radial plunger (or a reciprocating pumping element) 26. The input shaft (the pump shaft) of eccentric cam 24 is fixedly connected to the front end of countershaft 15 by means of a pin (not shown). Eccentric cam is operatively ~~accommodate~~ accommodated in a pump housing 25. Radial plunger 26 is slidably fitted into the cylinder defined in pump housing 25. Radial plunger 26 is permanently spring-loaded or biased toward the cam contour surface of eccentric cam 24 by means of a return spring 27. As shown in Figs. 2A and 2B, usually, the upper end of radial plunger 26 is in sliding-contact with the cam contour surface of eccentric cam 24, and thus radially reciprocates in synchronism with rotation of eccentric cam 24. During the suction stroke shown in Fig. 2B, working fluid (traction oil) is inducted from an oil pan into secondary oil pump 22 through an inlet port 28 and an inlet valve 29 owing to the upward stroke (viewing Fig. 2B) of radial plunger 26. As a matter of course, for the purpose of oil induction, inlet port 28 is formed in pump housing 25 in such a manner as to open at the underside of the oil level of the oil pan. During the discharge stroke shown in Fig. 2A, working fluid (traction oil) is discharged from a discharge port 30 formed in pump housing 25 through a discharge valve 31 into the oil pan owing to the downward stroke (viewing Fig. 2A) of radial plunger 26.

Please replace the paragraph starting at page 15, line 22 with the following:

As shown in Figs. 2A-2C, the upper end (the first end) of radial plunger 26 is formed as a sliding-contact surface that is usually in sliding-contact with the cam contour surface of

eccentric cam 24, whereas the lower end (the second end) of radial plunger 26 is formed as a relatively large-diameter pressure receiving flanged end. The relatively large-diameter flanged end of radial plunger 26 serves to define a plunger retracting chamber 23 located to be opposite to the pumping chamber that communicates with inlet port 28 via inlet valve 29 and communicates with discharge port 30 via discharge valve 31 and accommodates therein the return spring 27. Plunger retracting chamber 23 is defined in pump housing 25 in conjunction with the relatively large-diameter flanged end of radial plunger 26 (see Figs. 2A-2C). As will be fully described later, in the toroidal CVT of the embodiment, a modulated hydraulic pressure (P_C, P_L, Pt, P_P), which is constantly produced or generated by a hydraulic control circuit 36 (described later) and modulated from a discharge pressure from primary oil pump (the prime-mover driven oil pump) 21 during operation of the ~~primer~~ prime mover, is delivered or fed into plunger retracting chamber 23 of secondary oil pump 22.

Please replace the paragraph starting at page 16, line 11 with the following:

When the prime mover (engine 1) is stopped and conditioned in its inoperative state, there is no hydraulic pressure produced by primary oil pump 21. In this case, radial plunger 26 of secondary oil pump (output-rotation driven pump) 22 is brought into sliding-contact with the cam contour surface of eccentric cam 24 by way of the spring bias of return spring 27. Under this condition, the secondary oil pump system permits radial plunger 26 to reciprocate between the stroke positions shown in Figs. 2A and 2B by means of eccentric cam 24 that is rotated in response to input rotation transmitted from the road wheels through transmission output shaft 17 and gearset 16 to countershaft 15. The reciprocating motion of radial plunger 26 causes a pumping action. Conversely when the prime mover (engine 1) is running and conditioned in its operative state, there is a supply of hydraulic pressure, produced by primary oil pump 21, into plunger retracting chamber 23. As a result of this, as shown in Fig. 2C, radial plunger 26 of secondary oil pump 22 is retracted and positioned to be spaced apart from the cam contour surface of eccentric cam 24. That is, during operation of the prime mover, second oil pump 22 can be held at the inoperative state by maintaining the pumping element (plunger 26) in a spaced, contact-free relationship with the pump driving element (eccentric cam 24) by way of the modulated hydraulic pressure (P_C, P_L, Pt, P_P), which is constantly produced by hydraulic control circuit 36 (described hereunder) during operation

of the ~~primer~~ prime mover. With radial plunger 26 held at its retracted position during operation of the prime mover (engine 1), radial plunger 26 is not driven by means of eccentric cam 24 that is rotated in response to input rotation transmitted from the road wheels to countershaft 15. As discussed above, in the toroidal CVT of the embodiment, even if eccentric cam 24 is rotated in response to rotation of the road wheels, secondary oil pump 22 can be kept in the inoperative state during operation of the prime mover (engine 1).

Please replace the paragraph starting at page 23, line 11 with the following:

Forward/reverse changeover valve 33 is designed so that its valve spool 33a is permanently biased in the spring-loaded position (the axially downward position) by way of the spring bias of return spring 33b, thereby normally establishing fluid communication between output circuit 42 of forward/reverse changeover valve 33 and output circuit 40 of forward ratio control valve 37 and fluid communication between output circuit 43 of forward/reverse changeover valve 33 and output circuit 41 of forward ratio control valve 37. With forward/reverse changeover valve spool 33a kept at the spring-loaded position, the ratio changing control suited to the forward running mode is enabled via forward ratio control valve 37, while the ratio changing control suited to the reverse running mode is disabled. During the forward running mode, fluid communication between two ports 33c and 33d is blocked by way of the uppermost land of spool 33a, and whereby there is no supply of line pressure P_L from sub circuit 35 through ports 33c and 33d, and sub circuit 48 into reverse ratio control valve 38, thus inhibiting the reverse-running-mode ratio-changing operation from being wastefully erroneously executed by reverse ratio control valve 38, during the forward running mode. ~~On the contrary~~ Conversely, during the reverse running mode (during reverse rotation of countershaft 15), spool 33a of forward/reverse changeover valve 33 is kept at the retracted position (the reverse operating mode position) by means of reverse sensor 56, so as to establish fluid communication between circuits 42 and 44 and fluid communication between circuits 43 and 45. In addition to the above, with spool 33a held at the retracted position, fluid communication between ports 33c and 33d is established, thus enabling line pressure P_L to be supplied from sub circuit 35 through sub circuit 48 to reverse ratio control valve 38. Thus, with forward/reverse changeover valve spool 33a kept at the retracted position, the ratio changing control suited to the reverse running mode is enabled via reverse

ratio control valve 38. As previously described, reverse sensor 56 is mechanically linked to countershaft 15 to cause axial movement of spool 33a of forward/reverse changeover valve 33 towards the retracted position against the spring bias of return spring 33b in response to reverse rotation of countershaft 15. In other words, reverse ~~reveres~~ sensor 56 contains moving parts. Therefore, lubricating oil is delivered from sub circuit 48 via a check valve 49 to reverse sensor 56, in particular during the reverse running mode, for lubrication of moving parts of reverse sensor 56. An excessive drop in line pressure P_L in sub circuit 48 may exert a bad influence on the accuracy of the reverse-running-mode ratio changing control executed by reverse ratio control valve 38. Thus, an opening pressure of check valve 49 is set to a pressure level that never results in an excessive drop in line pressure P_L in sub circuit 48.

Please replace the paragraph starting at page 37, line 29 with the following:

As the hydraulic pressure (hereinafter is referred to as "plunger retracted-position holding pressure") to be supplied to plunger retracting chamber 23 for holding radial plunger 26 of secondary oil pump 22 at the retracted position (see Fig. 2C) at which radial plunger 26 is positioned to be spaced apart from the cam contour surface of eccentric cam 24, and constantly produced by primary oil pump 21 (the ~~primer~~ prime mover driven oil pump) during operation of the prime mover (engine 1), the toroidal CVT of the embodiment uses either one of four sorts of hydraulic pressures P_C, Pt, P_L, and P_P (see Fig. 4). Concretely, as the plunger retracted-position holding pressure, any ~~either~~ one of cooler pressure P_C in cooling circuit 74, torque converter pressure Pt in torque converter pressure circuit 65, line pressure P_L in line pressure circuit 61, and pilot pressure P_P in pilot pressure circuit 68 can be used. More concretely, in the secondary oil pump structure shown in Figs. 2A-2C, either one of outlet ports indicated by A, B, C, and D in the hydraulic circuit diagram of Fig. 4 is fluidly connected to plunger retracting chamber 23 shown in Figs. 2A-2C.

Please replace the paragraph starting at page 39, line 31 with the following:

For instance, when cooler pressure P_C in cooling circuit 74 is used as the previously-discussed plunger retracted-position holding pressure, that is constantly produced during operation of the ~~primer~~ prime mover (engine 1), that is, during primary oil pump 21, the working fluid of cooler pressure P_C can be rapidly drained when primary oil pump 21 is

shifted to the stopped state. The use of cooler pressure P_C is superior in enhanced operational response of secondary oil pump 22, and also enhances the accuracy of ratio changing control.